

AN INVESTIGATION OF SOME OF THE  
CHARACTERISTICS OF A JERK PUMP  
INJECTION SYSTEM FOR  
DIESEL ENGINES  
—•••••  
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An Investigation of Some of the Characteristics  
of a Jerk Pump Injection System for Diesel Engines

By

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B.S. (United States Naval Academy) 1930

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## TABLE OF CONTENTS

	Page
Title Page . . . . .	1
Table of Contents . . . . .	2
List of Illustrations . . . . .	3
Statement of Problem . . . . .	4
Apparatus and Test Procedure . . . . .	5
Discussion and Results . . . . .	11
Conclusions . . . . .	24
Acknowledgment . . . . .	26
Bibliography . . . . .	27

TABLE OF CONTENTS

Page

1	.....	1970-1971
2	.....	1971-1972
3	.....	1972-1973
4	.....	1973-1974
5	.....	1974-1975
6	.....	1975-1976
7	.....	1976-1977
8	.....	1977-1978
9	.....	1978-1979
10	.....	1979-1980
11	.....	1980-1981
12	.....	1981-1982
13	.....	1982-1983
14	.....	1983-1984
15	.....	1984-1985

## LIST OF ILLUSTRATIONS

	Page
Figure 1. General View of Apparatus .....	5
Figure 2. Pictures of Fuel Pump and Nozzle Valve, in cut-away section .....	7
Figures 3, 4 and 5, Schematic diagrams of Pump Lift Curve, Spray Apparatus, and Wiring Diagram .....	10
Figure 6. Curves showing "The Influence of Pump Speed and Nozzle Opening Pressure on Discharge Rate for Various Rack Settings ", for a pipe length of 30 inches .....	13
Figure 7. Curves showing "The Influence of Pump Speed and Rack Setting on Discharge Rate for Various Nozzle Opening Pressures ", for a pipe length of 15 inches .....	16
Figure 8. Curves showing "The Influence of Pump Speed and Rack Setting on Discharge Rate for Various Nozzle Opening Pressures ", for a pipe length of 30 inches .....	17
Figure 9. Penetration Pictures for a Rack Setting of .400	20
Figure 10. Penetration Pictures for a Rack Setting of .600	21

# LIST OF ILLUSTRATIONS

Figure 1, General view of apparatus .....	1
Figure 2, Diagram of flow and results of test, in one-way system .....	2
Figures 3, 4 and 5, Diagrams of flow in two-way system, and of flow in one-way system .....	3
Figure 6, Curves showing the influence of pump speed and back pressure on the flow rate for various lengths of the operating pressure, for a pipe length of 10 inches .....	4
Figure 7, Curves showing the influence of pump speed and back pressure on the flow rate for various lengths of the operating pressure, for a pipe length of 15 inches .....	5
Figure 8, Curves showing the influence of pump speed and back pressure on the flow rate for various lengths of the operating pressure, for a pipe length of 20 inches .....	6
Figure 9, Investigation of flow for a back pressure of 400 .....	7
Figure 10, Investigation of flow for a back pressure of 500 .....	8



## STATEMENT OF PROBLEM

The purpose of this investigation was to determine the influence of pump speed, nozzle opening pressure, rack setting, and length of line on the discharge rate of the injection valve in a jerk pump injection system for Diesel engines. Since some of the above variables may be controlled by an operator for any given system, it is felt that any correlation, mathematical or experimental, between these several variables would be of value to the designer of the injection system as well as to the Diesel operator.

In addition to obtaining a correlation between the above variables, FOR A PARTICULAR INSTALLATION, it appeared desirable to determine whether or not these variables might be changed with respect to one another to obtain performance which might be predicted, and which would result in better operation.

With these objectives in mind the problem was attacked from a purely experimental point of view and the results are recorded herein.

EXPERIMENTAL INVESTIGATION

The purpose of this investigation was to determine the influence of pump speed, water quality, and length of line on the efficiency of the injection pump. In a first series of tests (see Table I) the pump was operated at a constant speed of 1000 rpm. The water was of the same quality and the length of the line was varied. The results of these tests are shown in Table II. It is seen that the efficiency of the pump decreases as the length of the line increases. This is due to the friction loss in the line. The friction loss is proportional to the length of the line and to the square of the flow rate. The friction loss can be reduced by using a larger diameter pipe or by using a pump with a higher flow rate. The results of the tests show that the efficiency of the pump is about 70% when the length of the line is 100 ft. and about 50% when the length of the line is 200 ft.

In addition to determining the efficiency of the pump, the effect of water quality on the efficiency of the pump was also investigated. The water used in the tests was of the same quality as the water used in the tests of the first series. The results of these tests are shown in Table III. It is seen that the efficiency of the pump is about 70% when the water is of the same quality as the water used in the tests of the first series. This is due to the fact that the water is of the same quality. The results of the tests show that the efficiency of the pump is about 70% when the water is of the same quality as the water used in the tests of the first series.

With these results in mind the pump was tested at a variety of different speeds. The results of these tests are shown in Table IV. It is seen that the efficiency of the pump increases as the speed of the pump increases. This is due to the fact that the flow rate of the pump increases as the speed of the pump increases. The results of the tests show that the efficiency of the pump is about 70% when the speed of the pump is 1000 rpm and about 80% when the speed of the pump is 1500 rpm.



## APPARATUS AND TEST PROCEDURE

A general view of the apparatus is shown in Figure 1, and a schematic diagram of the same apparatus is shown in Figure 4. In the above two figures the same letters are used to designate the same parts. Referring to either Figure 1 or Figure 4:

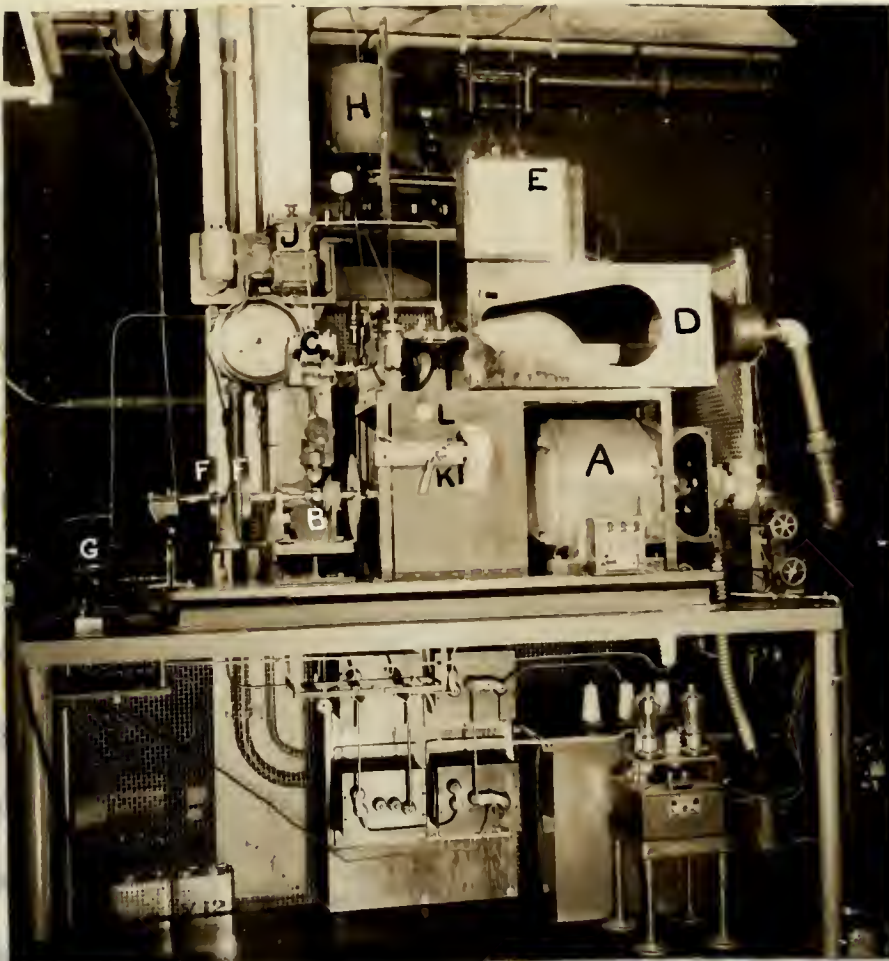


FIGURE 1.

"A" is a shunt motor having rheostatic speed control which drives the shaft to which is secured the cam "B", this cam in turn operates the plunger of the jerk pump "C". From the pump "C" the oil is discharged through a pipe line to the nozzle valve which is located in the box "D". A graduated scale is secured in box "D" as shown, by means of which the spray penetration may be determined. Mounted on

A general view of the structure is shown in Figure 1, and a  
 detailed diagram of the same apparatus is shown in Figure 2. In  
 the above two figures the main features are used to designate the  
 same parts. Indicated in Figure 1 are Figure 2.



FIGURE 1.

"A" is a short metal having resistance equal to that of the  
 the shell to which is secured the end "B", this end is then secured  
 the bottom of the pipe joint "C". From the pipe "D" the air is dis-  
 charged through a pipe line to the inside valve which is located in  
 the box "E". A graduated scale is secured to the "F" as shown, by  
 means of which the spring mechanism may be determined.



top of the box "D" is a stroboscopic neon light "E" which is made to illuminate the interior of the box "D" through a glass window in the top of "D". The time of the flashing of the neon light "E" may be made to occur at any desired angular shaft position by means of the graduated and variable rotary spark gap "F", one part of which is secured to the end of the motor shaft, while the variable part is mounted on the frame. A wiring diagram, Figure 5, shows the electrical connections between the spark gap "F" and the neon light "E". "H" is the oil reservoir located on one platform of a balance scale. Balance is obtained by placing weights of various magnitudes on the other platform; the instant of perfect balance being indicated by the flashing of a small neon light "L", Figure 1. This neon light is controlled by an electrical circuit through two small wires secured to either platform; the wires moving in and out of mercury baths, located under the platforms, as the balance changes position. From "H" the oil is delivered to the fuel pump "C" through the oil filter "J". It is to be noted that oil may be delivered to the pump "C" by gravity or under pressure by means of the variable speed pump "G". A pressure of 12 pounds per square inch was maintained at the suction side of the pump "C" throughout the investigation.

The pump used was a 10-millimeter "jerk-pump", having the conventional plunger scroll control. A cut-away picture of this type of pump is shown in Figure 2(B). The quantity of oil delivered by the pump may be varied by rotating the plunger "P" by means of the gear "G" which is secured to the plunger "P". Gear "G" is in turn

top of the box "1" is a cylindrical metal light "2" which is made  
 to illuminate the interior of the box "1" through a glass window  
 in the top of "1". The base of the cylinder of the light "2" is  
 only for the purpose of supporting the light "2" and is not  
 of the cylindrical and metallic nature of the light "2". The base of  
 the light "2" is made of the same material as the light "2", and the  
 part is mounted on the base. A small cylindrical metal light "3" is  
 the electrical connection between the light "2" and the base  
 light "3". "3" is the electrical connection between the light "2" and  
 the base. Balance is obtained by placing weights on various  
 positions on the other platform. The constant of the balance  
 being indicated by the flashing of a small lamp light "4". Figure 1.  
 This lamp light is controlled by an electrical circuit through two  
 small wires secured to either platform. The wires extend to and  
 are at various points, located under the platform, at the distance  
 between platforms. From "5" the oil is delivered to the fuel pump  
 "6" through an oil filter "7". It is to be noted that all oil is  
 delivered to the pump "6" by gravity or under pressure by means of  
 the electric power pump "8". A pressure of 15 pounds per square  
 inch was maintained at the various ends of the pump "6" throughout  
 the investigation.

The fuel used was a 10-40-40 motor "9" which is a  
 standard 10-40-40 motor control. A fuel-air mixture of 10:1  
 is used in figure 1(a). The quantity of oil delivered by  
 the pump was for testing in testing the pump "6" by means of the  
 pump "6" which is secured to the platform "7". Gas "9" is in test



actuated by means of an engaging rack, the position of which is controlled by the micrometer head "R", Figure 4. The end of the plunger "P" is actuated by the cam "B", Figure 4, by means of suitable linkage. The operation of the fuel pump "B", Figure 2, is as follows: as the plunger "P" moves from right to left, the space "E" becomes isolated when the flat plunger face reaches the left edge of the inlet channel "J". Further motion of "P" to the left causes the plunger to force oil from "E" past the check valve "V" until the left edge of the scroll space "S" reaches the right edge of the inlet channel "J". At this point the space "E" communicates directly with the inlet channel "J" through a small hole drilled from the plunger face to the scroll space "S", and the oil in space "E" is by-passed back to "J".

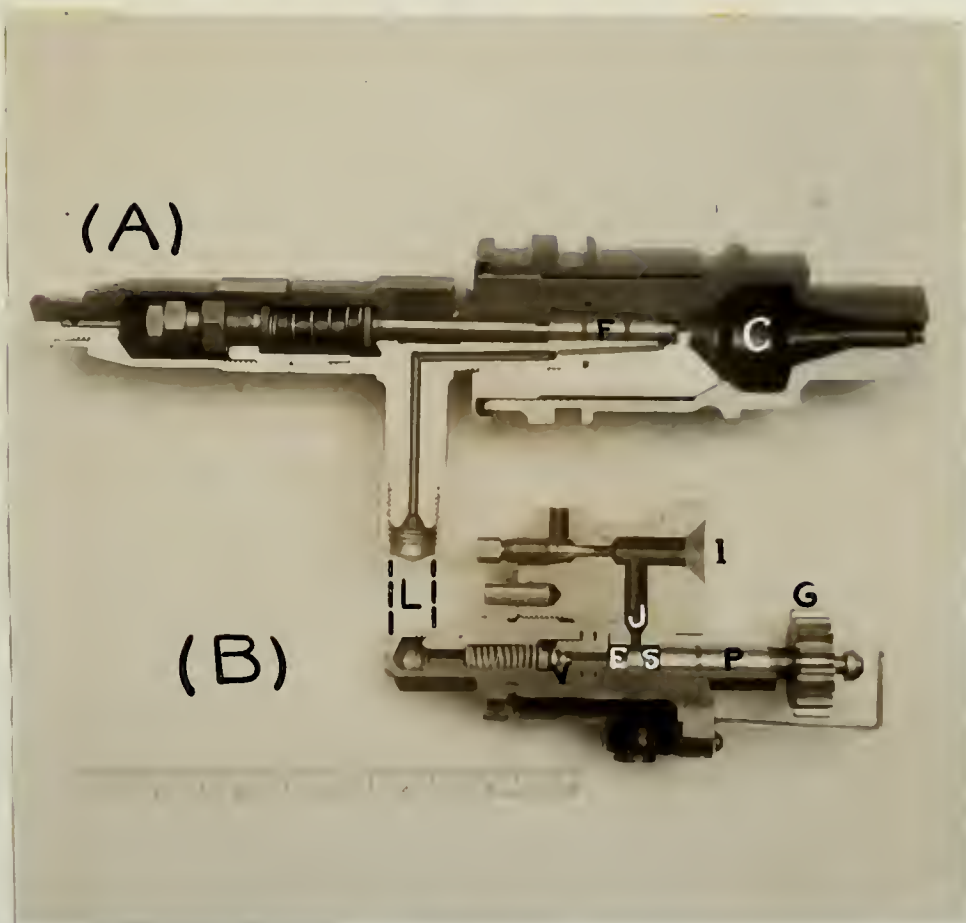


Figure 2.

entrance to each of the two pipes, the position of which is also  
 marked by the dimension lines  $2''$ , figure 4. The end of the pipes  
 $1''$  is indicated by the line  $1''$ , figure 4, by means of vertical lines.  
 The operation of the two pipes  $1''$ , figure 4, is as follows: as the  
 plunger  $1''$  moves from right to left, the space  $1''$  between the  
 ends of the two pipes (the space between the left edge of the left channel  
 $1''$ , figure 4, and the left corner of the plunger  $1''$ ) is  
 all then  $1''$  past the short pipe  $1''$  until the left edge of the  
 small space  $1''$  reaches the right edge of the left channel  $1''$ . At  
 this point the space  $1''$  communicates directly with the left channel  
 $1''$  through a small hole drilled from the plunger  $1''$  to the small  
 space  $1''$ , and the air in space  $1''$  is by-passed from  $1''$ .



Figure 2



Thus in this type of pump, delivery will always commence at the same angular shaft position, regardless of rack setting, whereas the angular shaft position at which oil ceases to be delivered will depend upon the setting of the rack.

Oil enters the pump Figure 2(B) at "I", is discharged past the check valve "V" from whence it is delivered to the nozzle valve, Figure 2(A) through the pipe line "L".

A Caterpillar Fuel Injection Valve designed for engines of  $5\frac{1}{4}$ " and  $5\frac{3}{4}$ " cylinder bore was used in this investigation. A picture of this valve, in cut-away section, is shown in Figure 2(A); the valve actually used, however, differs from the one shown in Figure 2(A) in that the pre-combustion chamber "C" was not used. The manufacturer's part designation for this valve is as listed below:

<u>Part Name</u>	<u>Part Number</u>	<u>Weight (Grams)</u>
Spray valve spring	1A6926	11.22
Spray valve spring stem	2A4684	16.07
Spray valve needle	2A4682	5.74

Other pertinent data pertaining to the nozzle valve are:

- (1) needle valve lift 0.007"; (2) needle valve stem diameter 0.039";
- (3) included angle between faces of valve seat =  $60^\circ$ ; (4) orifice length 0.118"; (5) orifice diameter 0.025"; (6) spring constant 771.2 lbs. per in. deflection; (7) clearances between "P" and "F" and their respective working surfaces = lap fit.

In obtaining the data necessary for plotting the curves shown

in Figure 1, 2 and 3, the adjustable spring of the injection valve was fixed at 100 lb. and the air in a small pump to cause the valve to open at the desired pressure. The pump used was equivalent of a constant volume and the number of pump strokes required for the delivery of a known weight of oil, as indicated by the balance and indicator light system, was recorded on the provided number "1". Figures 1 and 2. Several "runs" were made for each setting in order to obtain a series of data.

The spray operation photos shown in Figure 2 were taken at a pump speed of 800 R.P.M. and a tank setting of 100 (constant load). These in Figure 3 were taken at the same pump speed but for a tank setting of 200 (light load). These spray photos in both cases were 1/100 sec. per sq. in. Exposures were taken for every one degree regular about displacement, starting with the first stroke of the injection and continuing until past the point of cut-off. Since a speed of 800 R.P.M. is equivalent to one revolution in a tank of a second, the photos were given a time exposure of one-half of a second. Spray giving only one injection per stroke rather than a series of injections. The series of pictures represent several hundred injections rather than a time development of one injection.

The characteristic properties of the diesel fuel used were as follows:

Gravity.....	50.8 A.S.T. at 60°F.
Viscosity.....	28.7 A.S.T. at 100°F.
Specific Gravity.....	0.858 at 60°F.



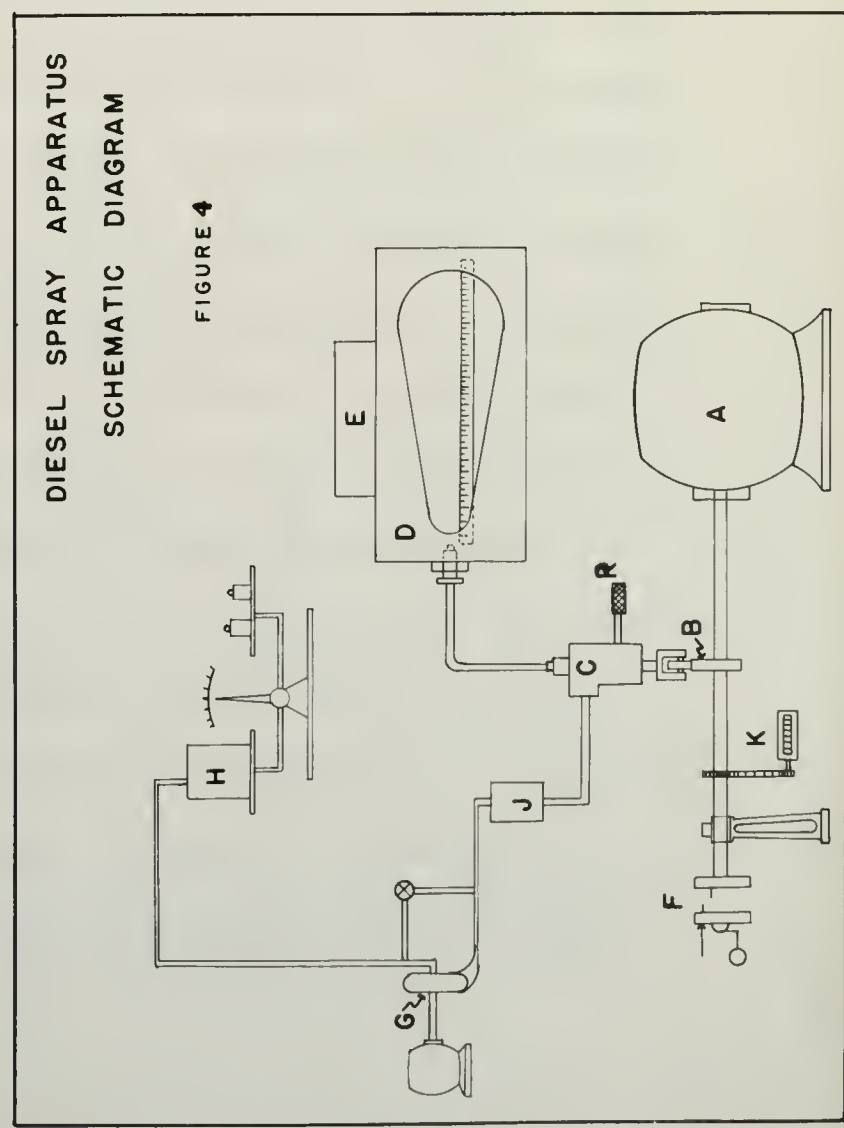
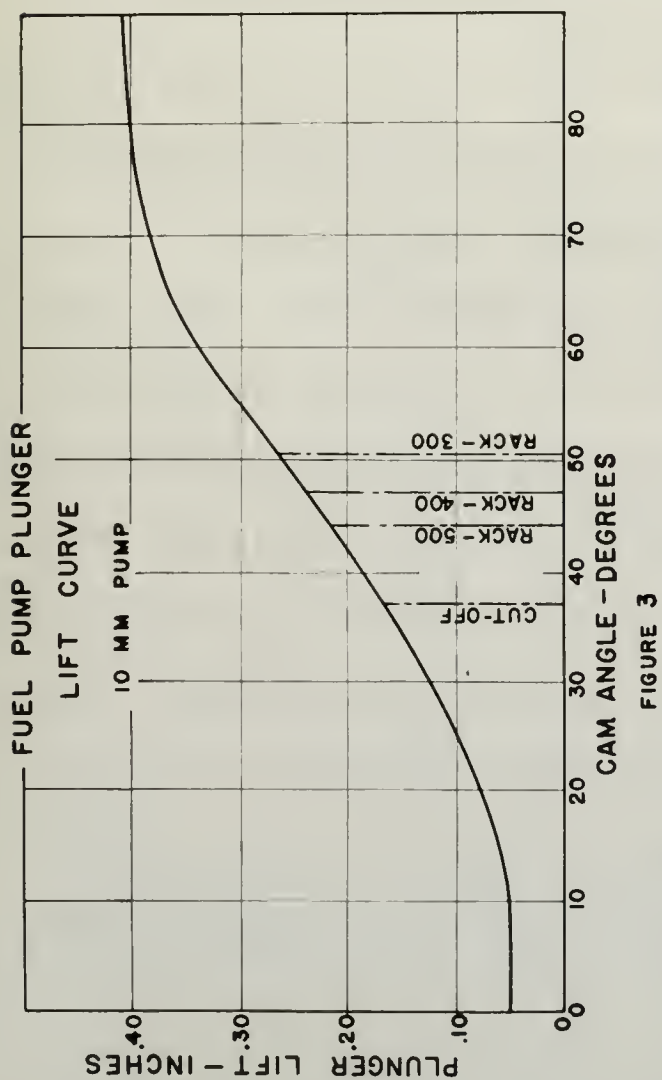


FIGURE 4

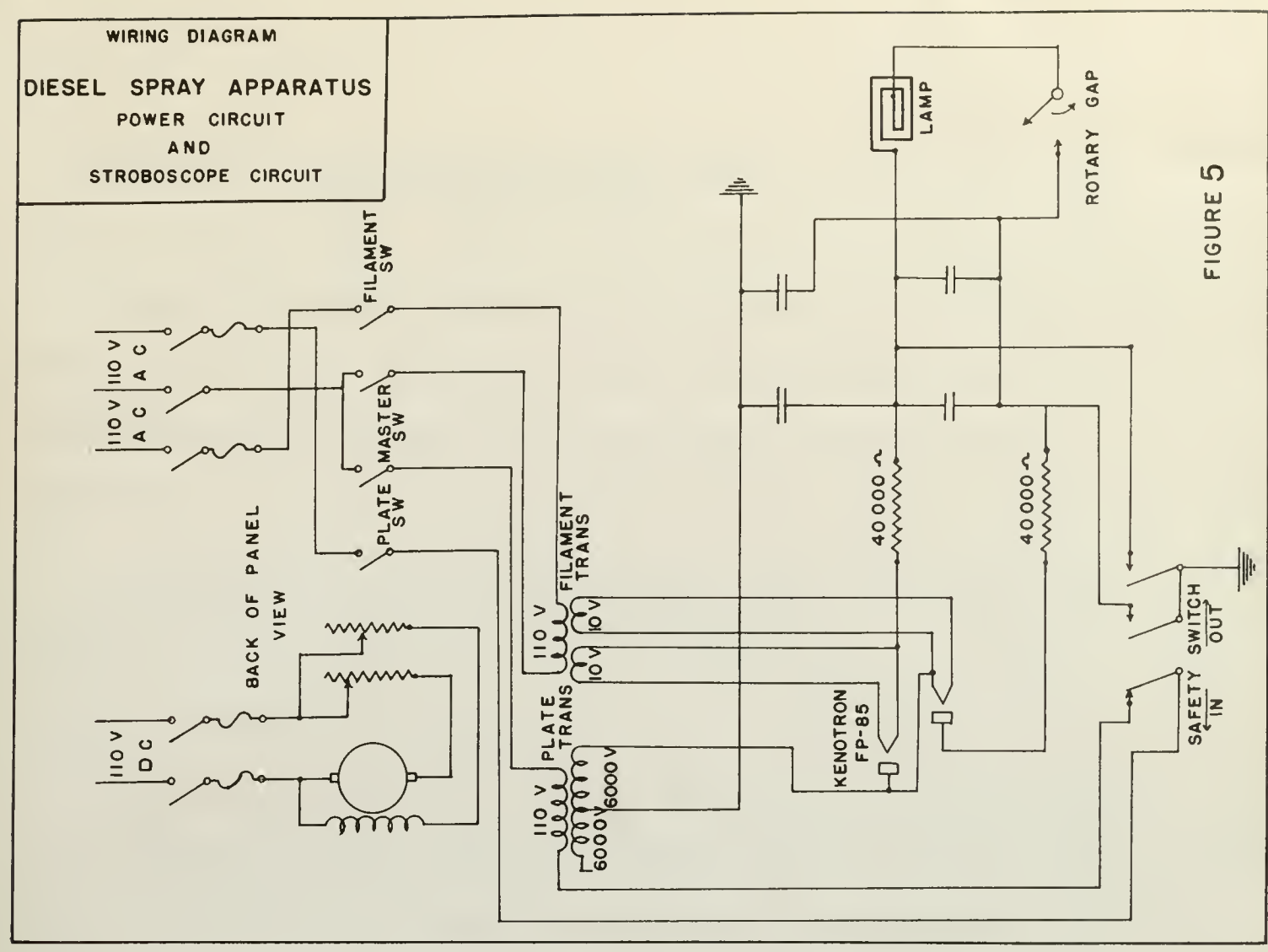


FIGURE 5





## DISCUSSION AND RESULTS

The quantity of oil which the fuel pump, Figure 2(B), should deliver for various rack settings, based on displacement data, was computed as follows and is shown by the plane surfaces A B C D in Figure 6:

Dial indicator readings were carefully taken of the cam contour for the various angular crank positions from which the lift curve, Figure 3, was constructed. Then the angular shaft position for cut-off (the point at which the fuel pump, Figure 2(B), began delivery) was obtained by manually unseating the check valve and observing the point at which the oil ceased to flow from the fuel pump; the oil to the suction side of the pump being maintained under pressure due to the gravity head from supply tank "H", see Figure 1. This angular shaft position was, as should be expected, the same for all rack settings. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the scroll position), was determined for each rack position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular shaft position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any rack setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any rack setting, the pump plunger area and the density

The quantity of oil which the pump (Figure 1) would deliver for various crank positions, based on displacement data, was computed as follows and is shown in the table between A & B on

Figure 2.

Dial indicator readings were carefully taken of the cam surface for the various angular crank positions from which the lift curve, Figure 3, was constructed. From the angular crank position the lift-off (the point at which the fuel pump, Figure 4(a), began delivery) was obtained by mentally measuring the crank angle and observing the point at which the oil ceased to flow from the fuel pump. The oil for the suction side of the pump being contained under pressure due to the gravity head from supply tank "B", see Figure 1. This angular crank position was, as should be expected, the same for all four strokes. Next, the angular position of the shaft at the point of release (that point at which the pump stopped delivery due to the supply position), was determined for each crank position. This was accomplished by again lifting the fuel pump check valve manually, and observing the angular crank position at which oil began to flow from the pump. These angular positions were then transferred to the lift curve, Figure 3, from which the effective pump stroke for any crank setting was obtained by subtracting the lift at the point of cut-off from the lift at the point of release. Knowing the effective pump stroke, for any crank setting, the pump chamber area and the density



of the Diesel oil, the weight of oil discharged per stroke was computed from the formula: weight per stroke = area of piston x stroke of piston x oil density. Since the area of the pump was .000846 square feet (plunger diameter 10 millimeters), and the density of the oil at 12 lbs. per sq. in. pressure and 74°F, was 62.284 x .8676 = 54.038 pounds per cubic foot, the weight of oil delivered per stroke was: .000846 x 54.038 x  $\frac{\text{lift}''}{12}$  = .00381 x lift'' (lbs.). The results of the above computations are as tabulated:

TABLE I

<u>Rack Setting</u>	<u>Cut Off</u>	<u>Re-lease</u>	<u>Lift at Cut Off</u>	<u>Lift at Release</u>	<u>Effective Pump Stroke</u>	<u>Calculated Wt. of oil per Stroke (lbs.)</u>
.150	376	56°	.175"	.305"	.130"	4.96 x 10 <sup>-4</sup>
.175	"	55°	"	.297"	.122"	4.65 x 10 <sup>-4</sup>
.200	"	54°	"	.290"	.115"	4.38 x 10 <sup>-4</sup>
.225	"	53°	"	.282"	.107"	4.08 x 10 <sup>-4</sup>
.250	"	52°	"	.274"	.099"	3.77 x 10 <sup>-4</sup>
.300	"	50.5°	"	.262"	.087"	3.315 x 10 <sup>-4</sup>
.350	"	49.25°	"	.253"	.078"	2.97 x 10 <sup>-4</sup>
.400	"	47.25°	"	.238"	.063"	2.40 x 10 <sup>-4</sup>
.450	"	46°	"	.227"	.052"	1.98 x 10 <sup>-4</sup>
.500	"	44°	"	.213"	.038"	1.45 x 10 <sup>-4</sup>
.550	"	42.5°	"	.202"	.027"	1.03 x 10 <sup>-4</sup>
.600	"	40°	"	.185"	.010"	0.381 x 10 <sup>-4</sup>

From Figure 6, it may be observed that the surface representing the oil actually discharged per stroke for any given rack setting

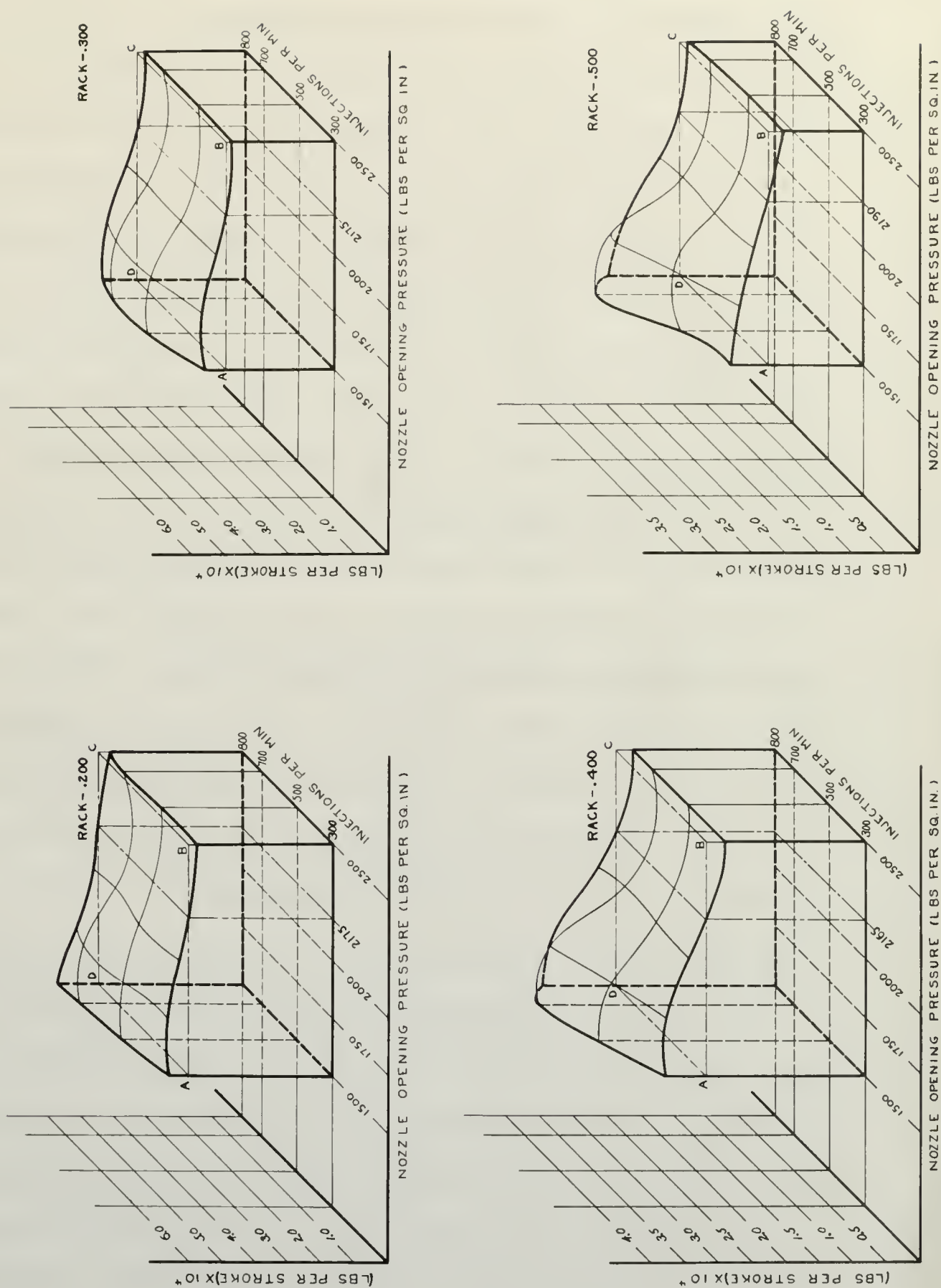
of the liquid oil, the weight of oil absorbed per atom was determined from the formula: weight oil absorbed = area of liquid x density of liquid x oil density. Since the area of the pump was 10.14 in<sup>2</sup>, the weight of oil absorbed was 10.14 x 1.05 x 0.85 = 9.07 g. The weight of oil delivered per atom was 9.07 g. The weight of oil delivered per atom was 9.07 g. The weight of oil delivered per atom was 9.07 g.

TABLE I

Weight of oil per stroke (g.)	Area of pump (in <sup>2</sup> )	Area of liquid (in <sup>2</sup> )	Area of oil (in <sup>2</sup> )	Area of oil (in <sup>2</sup> )	Area of oil (in <sup>2</sup> )
1.00 x 10 <sup>-4</sup>	1.00	1.00	1.00	1.00	1.00
1.10 x 10 <sup>-4</sup>	1.10	1.10	1.10	1.10	1.10
1.20 x 10 <sup>-4</sup>	1.20	1.20	1.20	1.20	1.20
1.30 x 10 <sup>-4</sup>	1.30	1.30	1.30	1.30	1.30
1.40 x 10 <sup>-4</sup>	1.40	1.40	1.40	1.40	1.40
1.50 x 10 <sup>-4</sup>	1.50	1.50	1.50	1.50	1.50
1.60 x 10 <sup>-4</sup>	1.60	1.60	1.60	1.60	1.60
1.70 x 10 <sup>-4</sup>	1.70	1.70	1.70	1.70	1.70
1.80 x 10 <sup>-4</sup>	1.80	1.80	1.80	1.80	1.80
1.90 x 10 <sup>-4</sup>	1.90	1.90	1.90	1.90	1.90
2.00 x 10 <sup>-4</sup>	2.00	2.00	2.00	2.00	2.00
2.10 x 10 <sup>-4</sup>	2.10	2.10	2.10	2.10	2.10
2.20 x 10 <sup>-4</sup>	2.20	2.20	2.20	2.20	2.20
2.30 x 10 <sup>-4</sup>	2.30	2.30	2.30	2.30	2.30
2.40 x 10 <sup>-4</sup>	2.40	2.40	2.40	2.40	2.40
2.50 x 10 <sup>-4</sup>	2.50	2.50	2.50	2.50	2.50
2.60 x 10 <sup>-4</sup>	2.60	2.60	2.60	2.60	2.60
2.70 x 10 <sup>-4</sup>	2.70	2.70	2.70	2.70	2.70
2.80 x 10 <sup>-4</sup>	2.80	2.80	2.80	2.80	2.80
2.90 x 10 <sup>-4</sup>	2.90	2.90	2.90	2.90	2.90
3.00 x 10 <sup>-4</sup>	3.00	3.00	3.00	3.00	3.00

From Table I, it can be observed that the values corresponding to the oil actually absorbed per stroke for each pump setting.

THE INFLUENCE OF PUMP SPEED AND NOZZLE  
OPENING PRESSURE  
ON DISCHARGE RATE FOR VARIOUS RACK SETTINGS  
PIPE LENGTH - 30 INCHES  
FIGURE 6







changes contour with both speed and pressure variations. Disregarding the dynamics of the system this surface should be coincident with the plane surface ABCD which represents the amount of oil which the pump should deliver based on displacement data. Obviously the planes A B C D in Figure 6 are horizontal as shown since the theoretical weight delivered per stroke is independent of all variables save rack setting. That the weight per stroke contour surface does not coincide with the plane surfaces A B C D Figure 6, is clearly shown for the rack settings considered (.200, .300, .400, .500). The fact that the pump actually delivers more (or less) oil than is shown by displacement computations may be explained in the following way.<sup>(1) (2) (6)\*</sup>

Once the check valve has been lifted off its seat, due to the oil pressure created by the plunger motion, oil will flow into the line and pressure waves will develop between the face of the pump plunger and the nozzle valve. When the pump reaches the point of release, the check valve will remain off its seat for an appreciable interval due to its inertia as well as to the friction force occasioned by the viscous drag of the oil flowing past the valve. Thus the next pressure wave, after release, reflected from the plunger face will force additional oil past the open check valve and into the fuel line. This action is possible, in spite of the fact that the plunger by-pass channel puts the pump chamber in direct communication with the suction side of the pump, since the by-pass channel area is so small that the pressure wave moving toward the plunger face will be reflected from the plunger face before it has driven any oil out of the chamber space into the suction line. After the check valve has been seated, pressure

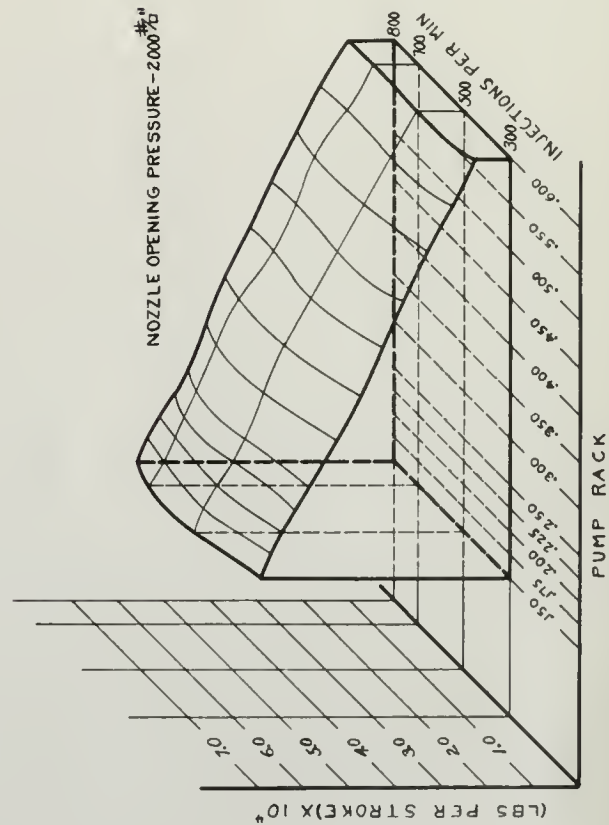
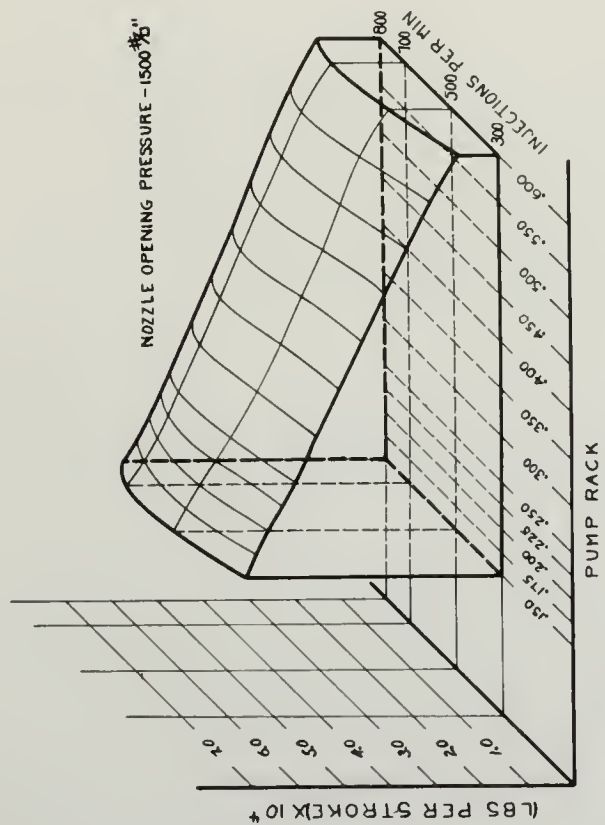
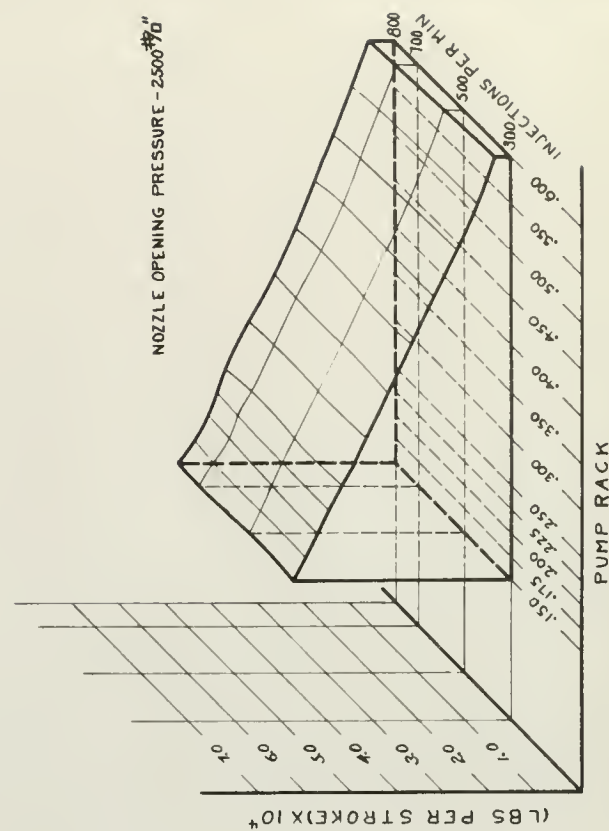
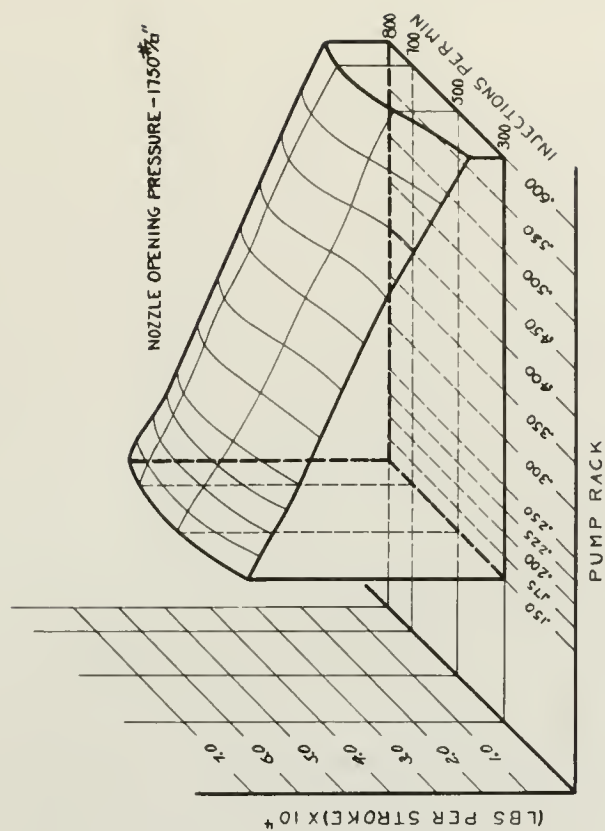
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\*Such designations refer to similar numbers in bibliography.



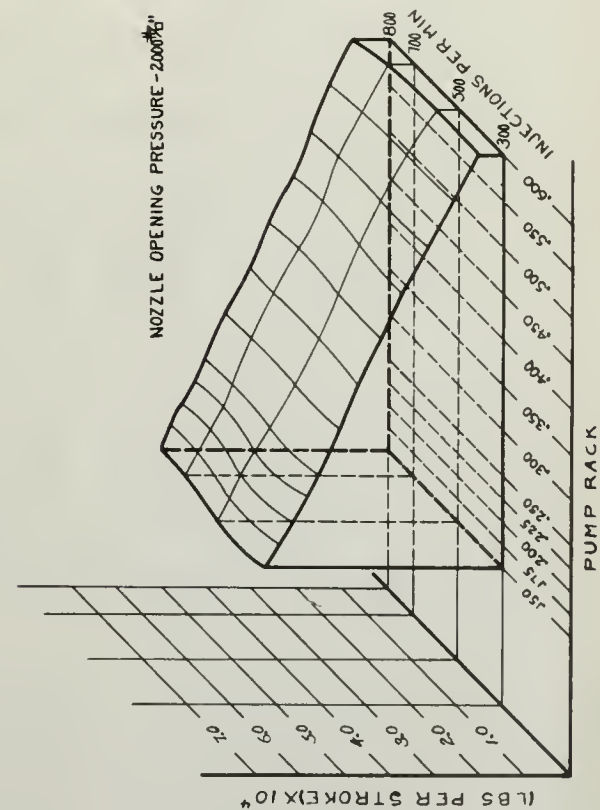
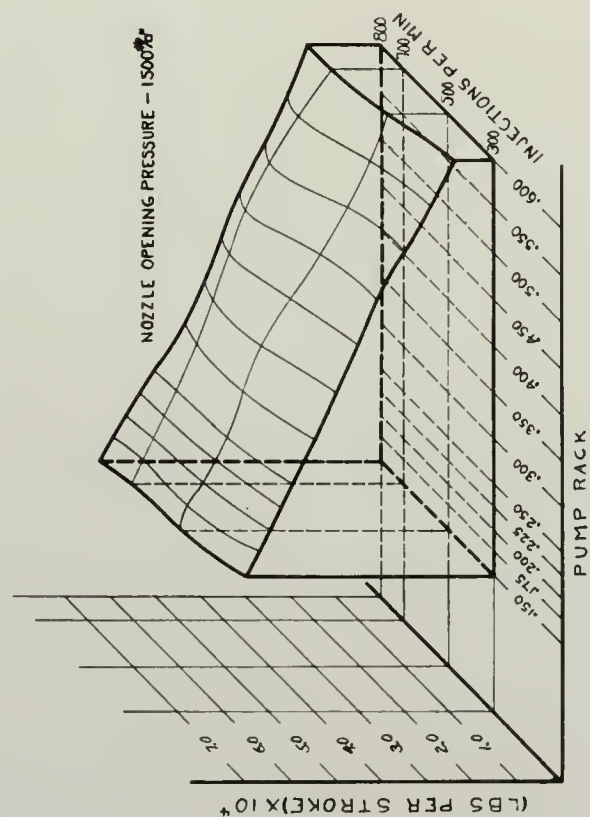
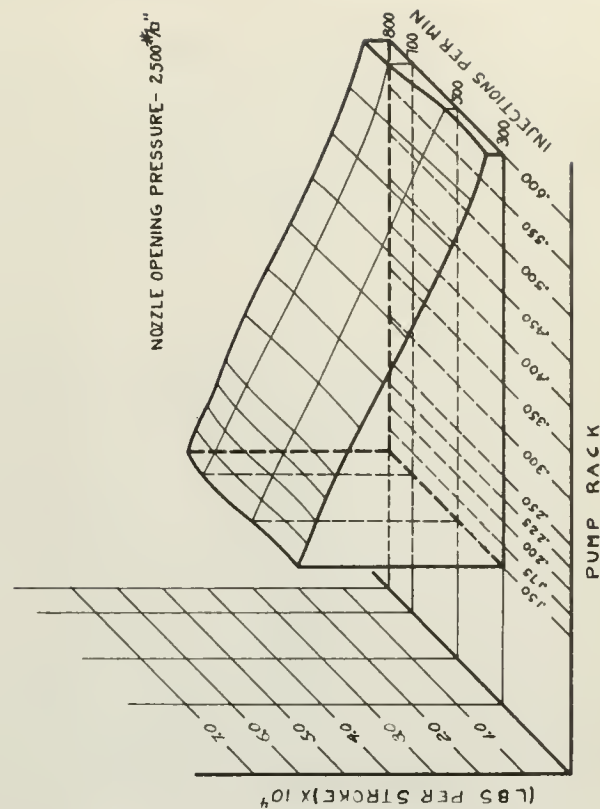
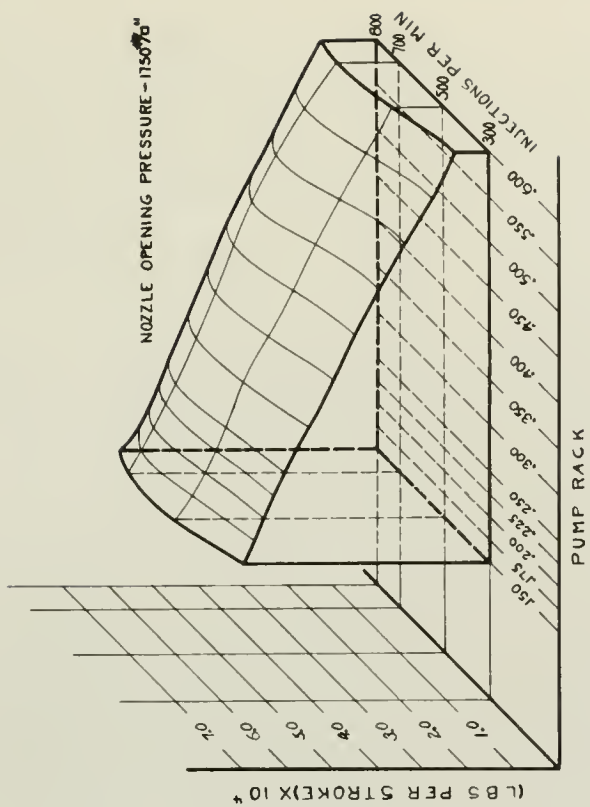


THE INFLUENCE OF PUMP SPEED AND RACK SETTING  
ON DISCHARGE RATE  
FOR VARIOUS NOZZLE OPENING PRESSURES  
PIPE LENGTH - 15 INCHES  
FIGURE 7





THE INFLUENCE OF PUMP SPEED AND RACK SETTING  
ON DISCHARGE RATE  
FOR VARIOUS NOZZLE OPENING PRESSURES  
PIPE LENGTH - 30 INCHES  
FIGURE 8





surfaces A B C D. The amount of oil actually delivered decreased with increasing nozzle opening pressures due to the greater force exerted by the oil to keep the check valve on its seat against the action of the pressure surges, and to the increased pump leakage occasioned by the higher discharge pressures.<sup>(5)</sup>

Figure 7 shows the influence of rack setting and speed on the weight of oil discharged per stroke for various nozzle opening pressures and a pipe length of 15 inches; Figure 8 shows a series of similar surfaces for a pipe length of 30 inches. Again it may be noted from these figures that as the nozzle opening pressure approaches 2175 lbs. per sq. in., the weight contour surface approaches plane surfaces; see Figures 7 and 8 for nozzle opening pressures of 2000 and 2500 lbs. per sq. in. These contour surfaces also clearly show the influence of pump speed changes on the quantity of oil discharged. Here it is to be noted that in general the weight of oil discharged increased with higher speed up to about 700 R.P.M.<sup>(3)</sup> while for the range between 700 -- 800 R.P.M., the weight of oil decreased;<sup>(5)</sup> this may be seen most clearly in Figures 7 and 8 and Table II for nozzle opening pressures of 1500 and 1750 lbs. per sq. in. For the nozzle opening pressures of 2000 and 2500 lbs. per sq. in. (in the neighborhood of 2175 lbs. per sq. in.) the influence of pump speed was not so marked.

Table II follows.



with increasing mass, the frequency of the sound waves  
 received by the ear is very low and the sound is  
 of the pressure range, and is the intensity of the  
 received by the higher frequency range.<sup>(1)</sup>

Figure 7 shows the influence of mass, weight and speed on the  
 weight of all distributed per stroke for various masses operating in  
 water and a type length of 10 inches; Figure 8 shows a series of  
 similar curves for a type length of 20 inches. Again it may be  
 noted from these figures that as the mass weight increases ap-  
 proximately 10% the weight of the water surface increases  
 about 10%; and as the mass weight increases 20% the  
 weight of the water surface increases about 20%. This shows that the  
 influence of mass weight on the sound is very small.  
 It is to be noted that in general the weight of all  
 distributed increases with higher speed up to about 100 ft./sec.<sup>(2)</sup>  
 while for the range between 100 -- 200 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 100 and 200 ft./sec.  
 For the mass operating between 200 and 300 ft./sec., the weight of all  
 distributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 200 and 300 ft./sec.  
 For the mass operating between 300 and 400 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 300 and 400 ft./sec.  
 For the mass operating between 400 and 500 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 400 and 500 ft./sec.  
 For the mass operating between 500 and 600 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 500 and 600 ft./sec.  
 For the mass operating between 600 and 700 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 600 and 700 ft./sec.  
 For the mass operating between 700 and 800 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 700 and 800 ft./sec.  
 For the mass operating between 800 and 900 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 800 and 900 ft./sec.  
 For the mass operating between 900 and 1000 ft./sec., the weight of all dis-  
 tributed is nearly constant. This may be seen clearly in Figure 7 and 8 and  
 Table II for mass operating between 900 and 1000 ft./sec.

Table II (Continued)

TABLE II

Rack Setting	Pump Speed	ACTUAL DISCHARGE (LBS. PER STROKE X 10 <sup>0</sup> ); Pipe Length 30".			
		Nozzle Opening	Pressure (Lbs. Per	Sq. In.)	
		1500	1750	2000	2500
.200	300	498	505	467	418
	500	541	535	485	412
	700	556	550	450	424
	800	568	508	457	403
.300	300	394	404	366	318
	500	459	452	365	311
	700	460	457	359	312
	800	435	422	361	319
.400	300	303	304	269	216
	500	352	337	265	216
	700	388	373	257	217
	800	355	336	270	217
.500	300	202	191	166	122
	500	226	236	173	123
	700	300	279	167	128
	800	254	228	174	132

The fact that the quantity of oil increased for increased pump speed up to 700 R.P.M., and then decreased for further speed increases may be explained as follows: higher pump speeds give higher plunger speed, and hence impart greater velocities to the oil being discharged from the pump; this increase in kinetic energy causes a greater quantity of oil to flow past the pump check valve after release and before the check valve has become seated. As the pump speed increased beyond a certain value, however, the volumetric efficiency of the pump decreased since oil could not flow into the pump chamber fast enough to completely fill it, and the quantity of oil discharged decreased.

A comparison of Figures 7 and 8 shows pipe length, for the two

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1991	1992	1993	1994	1995	
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the fact that the quantity of oil contained in the barrel was  
found up to 100 lbs. and then decreased the higher the barrel  
was as explained as follows: As the barrel was tilted  
upward, and hence higher position relative to the oil being displaced  
from the barrel; this increase in height caused a greater quantity  
of oil to flow into the barrel with each stroke and before the  
stroke when the barrel rested. As the barrel was tilted upward a  
greater volume, however, the volume of oil in the barrel  
increased and the barrel was tilted upward to completely  
fill it, and the quantity of oil displaced decreased.

A comparison of 1/250000 7 and 8 shows little difference.





Figure 9

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds

Nozzle Opening Pressure, 1750 lbs. per sq. in.

Pump Speed, 600 R.P.M.

Length of Pipe Line, 15 inches

Rack Setting, .400 (2.40 x 10<sup>4</sup> lbs. of oil per stroke)

Chamber Pressure, Atmospheric





Figure 10

SPRAY DEVELOPMENT FOR AVERAGE OF SEVERAL HUNDRED INJECTIONS.

Time Between Frames, .000278 seconds  
 Nozzle Opening Pressure, 1750 lbs. per sq. in.  
 Pump Speed, 600 R.P.M.  
 Length of Pipe Line, 15 inches  
 Rack Setting, .600 (.381 x 10<sup>4</sup> lbs. of oil per stroke)  
 Chamber Pressure, Atmospheric.





lengths investigated, to have little influence.<sup>(3)</sup> For nozzle opening pressures of 1750 and 2500 lbs. per sq. in., the two pipe lengths gave almost identical contour surfaces, while for nozzle opening pressures of 1500 and 2000 lbs. per sq. in. some irregularities between the contour surfaces may be noted. These irregularities, as may be seen, occurring for pump speeds of 500 R.P.M. and above.

A comparison of Figures 9 and 10 show the influence of rack setting on penetration at a pump speed of 600 R.P.M., and a nozzle opening pressure of 1750 lbs. per sq. in., when discharging against atmospheric pressure. Figure 9 is for a rack setting of .400, while Figure 10 is for a rack setting of .600. For both rack settings, the pictures marked 1 were taken at two degrees of pump shaft angular displacement after the point of injection was observed which is equivalent to one eighteen-hundredth of a second. Thereafter the pictures were taken in succession at one degree pump shaft intervals (one thirty-six-hundredth of a second). In Figure 9, evidence of secondary discharges may be seen in pictures 4, 5, 13, 14, 15 and 17.<sup>(4)</sup> Maximum penetration of 16.5 inches is shown in picture 12 and cut off in picture 13. The depth of penetration is quite uniform up to the point of cut off, picture 13. It is interesting to note at this point that Figure 9 shows a definite injection period over an interval of 13 degrees angular pump shaft displacement, whereas the table on page 12 for a rack setting of .400 shows the injection period to be only 10.25 degrees. This point again established the fact that the pump actually discharges more oil under certain conditions than is theoretically possible.

The first of these is the fact that the  
 pressure of 1700 and 1800 lbs. per sq. in. is  
 not a constant value, but varies with the  
 position of the piston. The pressure is  
 highest at the top of the cylinder and  
 lowest at the bottom. This is due to the  
 fact that the weight of the gas is not  
 negligible. The pressure at the top of the  
 cylinder is about 1800 lbs. per sq. in. and  
 at the bottom it is about 1700 lbs. per  
 sq. in. The difference is about 100 lbs.

The second of these is the fact that the  
 pressure of 1700 and 1800 lbs. per sq. in.  
 is not a constant value, but varies with the  
 position of the piston. The pressure is  
 highest at the top of the cylinder and  
 lowest at the bottom. This is due to the  
 fact that the weight of the gas is not  
 negligible. The pressure at the top of the  
 cylinder is about 1800 lbs. per sq. in. and  
 at the bottom it is about 1700 lbs. per  
 sq. in. The difference is about 100 lbs.

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 position of the piston. The pressure is  
 highest at the top of the cylinder and  
 lowest at the bottom. This is due to the  
 fact that the weight of the gas is not  
 negligible. The pressure at the top of the  
 cylinder is about 1800 lbs. per sq. in. and  
 at the bottom it is about 1700 lbs. per  
 sq. in. The difference is about 100 lbs.



In Figure 10 it may be readily seen that the spray penetration is quite irregular. Secondary discharges may be noted in pictures 3, 4, 5, 6, 7, 9 and 10. Cut off has taken place in picture 7, and here the maximum penetration was about 10 inches. Again it may be noted in Figure 10 that injection took place over a pump shaft interval of 8 degrees whereas the table on page 12, for a rack setting of .600, shows this interval to be only 3 degrees.



## CONCLUSIONS

1. From Figure 6 it may be concluded that for this system there exists a certain nozzle opening pressure for which a linear relation exists between weight of fuel discharged per stroke and rack setting, and further that this linear relation is independent of speed for the range investigated; that is to say, for a pump speed from 300 -- 800 R.P.M. and a nozzle opening pressure of from 1500 -- 2500 pounds per sq. in. In this case the nozzle opening pressure required to give this linear relation was found to be about 2175 pounds per sq. in.

It appears reasonable to suppose that a similar nozzle opening pressure will exist, and may be found, for any injection system of the type investigated.

2. A comparison of Figures 7 and 8 shows that for the pipe lengths investigated there was little or no change in the weight of oil delivered per stroke for the two pipe lengths.

3. Figures 6, 7 and 8 show definitely that the quantity of oil discharged per stroke decreases with increase in the nozzle opening pressure.

4. Figures 6, 7 and 8 show that when operating at a nozzle opening pressure other than 2175 pounds per sq. in., as mentioned in 1 above, the weight of oil discharged per stroke will vary with pump speed, and the greater the departure of the pressure





from this value of nozzle opening pressure, the greater will become the influence of pump speed. Generally speaking, the quantity of oil delivered per stroke, for the system investigated, increased with higher pump speed up to about 700 R.P.M., and further speed increase caused the quantity of oil discharged to decrease.

5. The quantity of oil delivered per stroke follows in a general way the rack setting, as seen from Figures 7 and 8, but this relation is not linear unless the nozzle is set to open at the proper spring setting (2175 pounds per sq. in. for this system); see Figure 6.

6. For a moderate load (rack setting of .400), it may be noted from Figure 9 that the spray penetration is reasonably uniform and that there is little evidence of secondary discharges. However, for light loads (rack setting of .600), it may be seen from Figure 10 that the spray penetration is irregular and that there is much evidence of secondary discharges.

The following are the names of the persons who have been identified as having been involved in the activities of the group:

[The rest of the page contains several lines of extremely faint, illegible text.]

3. The quantity of oil delivered per annum follows in a

3. For a moderate load (load rating of 400), it may be stated from Figure 3 that the epoxy resin is reasonably uniform and that there is little evidence of secondary shrinkage. However, for light loads (load rating of 1,000), it may be seen from Figure 11 that the epoxy penetration is irregular and that there is much evidence of secondary shrinkage.



### ACKNOWLEDGMENT

The author desires especially to thank Professor C. J. Vogt of the Department of Mechanical Engineering, University of California, who started this investigation and who has given untiringly and cheerfully of both his time and wisdom in order that this problem might be carried forward.

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Finally, I desire to thank members of the W. P. A. Project in the Department of Mechanical Engineering for their work in constructing the experimental station and making necessary alterations.

APPENDIX

The author has been especially fortunate in the selection of the Department of Industrial Engineering, University of California, for the study of this problem and the study of the life and work of the man who has been the most influential in the development of this problem.

There are also several other factors which are mentioned in the study of the life and work of the man who has been the most influential in the development of this problem. The study of the life and work of the man who has been the most influential in the development of this problem is a study of the life and work of the man who has been the most influential in the development of this problem.

Finally, I desire to thank the many people who have helped me in the study of the life and work of the man who has been the most influential in the development of this problem. The study of the life and work of the man who has been the most influential in the development of this problem is a study of the life and work of the man who has been the most influential in the development of this problem.

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